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FPD1H04/P-GB HK/JR/IP

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0320023.5 27 AUG 2003

3. Full name, address and postcode of the or of each applicant *(underline all surnames)*

Freepower Ltd.
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SP2 0SE
United Kingdom

Patents ADP number *(if you know it)*

8700726001

If the applicant is a corporate body, give the country/state of its incorporation

England A/L 22/3/04

4. Title of the invention

Bearing

5. Name of your agent *(if you have one)*

"Address for service" in the United Kingdom to which all correspondence should be sent
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51/77
2/3/4
Am.

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Country	Priority application number *(if you know it)*	Date of filing *(day / month / year)*

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Number of earlier UK application	Date of filing *(day / month / year)*

8. Is a Patents Form 7/77 (Statement of inventorship and of right to grant of a patent) required in support of this request?

YES

Answer YES if:
a) any applicant named in part 3 is not an inventor, or
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Continuation sheets of this form	
Description	12
Claim(s)	2
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Statement of inventorship and right to grant of a patent (Patents Form 7/77)

Request for a preliminary examination and search (Patents Form 9/77)

Request for a substantive examination (Patents Form 10/77)

Any other documents (please specify)

11. I/We request the grant of a patent on the basis of this application.

Signature(s) Hammonds

Date 27 August 2003

12. Name, daytime telephone number and e-mail address, if any, of person to contact in the United Kingdom

Kathleen Harris
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Applicant: Freepower Ltd
Attorney'ref: FP01H04/P-GB

DUPLICATE

Bearing

The present invention relates to rotary machine componentry, and more particularly relates to a multidirectional bearing.

There are many forms of bearing conventionally used in industry, for example journal bearings to support a shaft while it rotates in a rotating machine, such as a motor or alternator. For example, US-A-4,500,143 discloses a clearance control device in an inter-shaft turbojet engine bearing journal placed between a high pressure trunnion and a low pressure trunnion and including inserted between an inner ring and an outer ring, the high pressure trunnion including a series of longitudinal grooves and a series of longitudinal holes which are supplied with oil from an intake and an annular collector in the low pressure trunnion, from which the oil moves by centrifugation through radial holes into an annular recess in the downstream tightening nut of the outer ring, which in turn communicates with the grooves through holes formed in the nut. Clearance in the bearing journal is controlled by cooling the high-pressure trunnion at the maximum operating speeds of the turbine via circulating oil. By virtue of an appropriate circuit, the oil supplied by the collector lubricates the bearing and cools the roller bearing journal.

In addition, numerous thrust bearings are known. For example US-B-5862666 discloses a modified turbine engine design in which control of the forward load on a thrust bearing is improved, while reducing or eliminating the need for the use of a balance piston is disclosed. The turbine engine has a longitudinal axis and comprises, a compressor for pressurizing gases in the engine. The impeller has an impeller rear face and a tip. The engine further comprises a shaft mounted to a thrust bearing for rotation about the axis. The impeller is fixed to the shaft for rotation therewith. A cavity fixed within the engine casing is defined at least by the impeller rear face and is in communication with the pressurized gases at the impeller tip.

A problem arises in the lack of availability of bearing systems for compact scale rotating machinery. There is a need for such devices for supporting the shaft of a rotating component that is rotating at high speed. Moreover, a problem is that of providing a bearing system that operates as both a journal bearing and a thrust bearing in small-scale machinery. Bearings of this type must also be robust and reliable, so that they can be employed in systems operating 24 hours a day, seven days a week for extended periods (and have a life expectancy of the order of five years or more).

The present invention provides a bearing for supporting a shaft rotatable about an axis and at least partially disposed within a housing, comprising: a bearing member, fixedly attached to the housing and having a first bearing surface, opposite a second bearing surface on the shaft, said first and second bearing surfaces extending generally transverse to the axis, and a cylindrical internal channel defining a third bearing surface extending generally parallel to the axis and disposed opposite a fourth bearing surface on the shaft, the bearing member including conduits adapted to convey lubricating fluid into at least the space third and fourth bearing surfaces.

Preferably, the bearing member has, on the end thereof opposite the first bearing surface, a fifth bearing surface extending generally transverse to the axis.

Preferably, the bearing member has a generally T-shaped cross-section. Preferably, the first surface on the bearing element is defined by a raised annular surface on the top of the 'T' extending partially between the inner radial limit and the outer radial limit of the bearing member. Preferably, a plurality of elongate first recesses are provided extending radially in the first surface, thereby facilitating flow of lubricant fluid to the space opposite the first surface. Preferably, the first recesses extend partially between the inner radial limit and the outer radial limit of the first surface.

Preferably, a plurality of elongate second recesses are provided extending radially in the fifth surface, thereby facilitating flow of lubricant fluid to the space opposite the fourth surface. Preferably, the second recesses extend partially between the inner radial limit and the outer radial limit of the fifth surface.

Preferably, at a point between the opposite ends of the elongate part of the 'T'-shaped bearing member, a circumferential recess is defined in the surface at the outer radial limit of the bearing member. Preferably, a plurality of first lubrication channels are provided, extending radially between the circumferential recess and the inner radial limit of the bearing member, thereby permitting flow of lubricant fluid between the exterior of the bearing member and the internal cylindrical channel.

Preferably, the bearing member includes a plurality of second lubrication channels, each channel extending axially between a first elongate recess on the first surface and a respective opposite second elongate recess on the fifth surface.

Preferably, the number of first and/or second elongate recesses is between 2 and 8, and preferably 6.

Preferably, the number of second lubrication channels is between 2 and 8.

The bearing preferably further includes a washer, wherein, in use, one surface of the washer abuts the fifth surface of the bearing member and the other surface of the washer is adapted to abut a corresponding surface of a drive element, for example a turbine.

The invention further provides an energy recovery system, for extracting energy from a source of waste heat, the system being a closed system with a circulating working fluid, comprising a heat exchanger, an electromechanical conversion unit, a cooling system and a turbine unit, the heat exchanger supplying, in use, the working fluid to said turbine unit as a gas, wherein the turbine unit is mechanically coupled to the electromechanical conversion unit via a shaft, the shaft being supported by a bearing according to any of the appended claims.

Preferably, the system further includes a secondary working fluid supply line from the cooling system to the bearing whereby working fluid is supplied to the exterior of the bearing member, thereby providing the lubricant fluid for said bearing. Preferably, the working fluid is supplied to the bearing as a liquid.

An advantage of the present invention is that it provides a bearing that is compact in scale. Another advantage is that it is capable of acting as both a journal bearing and a thrust bearing. In certain embodiments, and advantage is that lubrication is provided by the working fluid, and no separate lubricant supply is needed.

The present invention will now be described, by way of example, with reference to the accompanying drawings in which:

Figure 1 is shows (a) schematic overview of an energy recovery system in accordance with one aspect of the invention, and (b) intermediate electronics modifying the output of the alternator;
Figure 2 is a schematic illustration of the derivation of one source of waste in one aspect of the invention;
Figure 3 illustrates in more detail the turbine unit and alternator of Fig. 1;
Figure 4 is an enlarged view of the turbine bearing in Fig. 3;
Figure 5 shows in more detail the bearing member employed in the bearing in Fig. 4, indicating fluid flows; and
Figure 6 illustrates an alternative (magnetic) coupling of the turbine unit and alternator of Fig. 1, in another aspect of the invention.

Turning to the drawings, Fig. 1(a) is a schematic overview of an energy recovery system in accordance with one aspect of the invention. A main heat exchanger 102 has at least one source fluid inlet 104 through which it receives a heated source fluid incorporating the thermal energy that is sought to be recovered by the system. The temperature of the source fluid upon entering the main heat exchanger 102 is designated t1.

The main heat exchanger 102 may be driven by any source of heat, and examples of the source fluid include hot air, steam, hot oil, exhaust gases from engines, manufacturing process waste hot fluid, etc. Alternatively, the heat source may be solar thermal energy that heats a suitable fluid (e.g. heat transfer oil) that forms the source fluid for the main heat exchanger 102.

Referring briefly to Fig. 2, this is a schematic illustration of the derivation of one source of waste in one aspect of the invention: an important example of wasted energy is the ubiquitous internal combustion engine, be it petrol, diesel or gas fuelled, reciprocating or turbine. The best simple cycle fossil fuelled engine (other than very large power stations or marine engines) will be between 35-40% efficient: this means that 60-65% of the energy from the fuel used to drive the engine is lost as waste heat.

Returning to Fig. 1(a), the source fluid exits the main heat exchanger 102, at a reduced temperature t2, via at least one source fluid outlet 106.

The main heat exchanger 102, which is suitably of the cross counter flow type, also has a working fluid inlet 108 and working fluid outlet 110, through which it receives (as a liquid at temperature t3) and despatches (at temperature t4), respectively, the working fluid of the system. The working fluid, which is heated and vapourised within the main heat exchanger 102, is carefully chosen so that its thermodynamic and chemical properties are suitable to the system design and the operational temperatures and pressures. In one embodiment, the working fluid is HFE-7100.

After exit from the working fluid outlet 110 of the main heat exchanger 102, the gaseous working fluid flows in the direction of arrows A to the turbine inlet 112 of turbine unit 114. The working fluid arrives at the turbine unit 114 at pressure p1, loses heat and pressure in driving the turbine (not shown) mounted on turbine shaft 116 within the turbine unit 114, and exits the turbine unit 114 via turbine outlets 118 at pressure p2, which is substantially lower than p1. In one embodiment, the pressure p1 is 11.5 bar absolute and the pressure p2 is 1.0 bar absolute.

In one embodiment, the turbine shaft 116 is mounted on a bearing (not shown) and is mechanically coupled to an alternator 120, e.g. the turbine and alternator armature (not shown) are mounted on a common shaft 116. In this way, high-speed rotation of the turbine shaft 116 causes electrical energy to be generated in the alternator 120, the consequent voltage appearing at the alternator output 122. The coupling of the turbine shaft 116 to the alternator 120 is described further hereinbelow with reference to Figs 3 to 5.

After exit from the turbine outlets 118, the working fluid travels in the direction of arrows B to inlet 124 of a second heat exchanger 126, which acts as a preheater of the working fluid using the turbine exhaust. The working fluid is therefore input to the second heat exchanger 126 at temperature t5 and exits via outlet 128 at a lower temperature t6. At the same time, the second heat exchanger receives another flow of working fluid (in the direction of arrows C), below its boiling point and in liquid form, via inlet 130 at temperature t7. In the second heat exchanger 126, thermal energy is transferred to the flow of working fluid arriving at inlet 130, the working fluid exits via outlet 132 at temperature t3, and flows (in the direction of arrows D) to the inlet 108 of the main heat exchanger 102.

The system also includes a condensing unit (or water cooler) 134, in which cold water arrives via inlet 136 and exits via outlet 138. In operation, working fluid from the second heat exchanger 126, flowing in the direction of arrow E, arrives in the condensing unit 134 via inlet 140, is cooled and condensed into a liquid in the condensing unit 134, and then departs via outlet 142. This liquid working fluid (at temperature t7), is forced by pump 144 via valve 146 in the direction of arrows C and forms the second supply of working fluid arriving at second heat exchanger 126, to begin the cycle all over

again. In one embodiment, a separate fluid line 160 delivers liquid working fluid to the bearing coupling the turbine unit 114 and the alternator 120, for lubrication.

Thus, the system operates on a Rankine cycle and is sealed, so that there is no escape or consumption of the working fluid, which simply cycles through its various phases.

In one embodiment, the system includes a control system 150, to control the power output by the system. Most existing Rankine cycle machines are low speed units with synchronous alternators, running at the same frequency as the grid supply. Turbine speed and power control is generally by valves to bypass the turbine. However, the system according to one aspect of the present invention employs a high-speed alternator 120, and a power-conditioning unit is preferably used to convert the high frequency alternator output to mains frequency.

More specifically, the control system includes intermediate electronics 151, a power conditioning unit (PCU) 152 and a controller 154. The power output by the alternator 120 at outputs 122 is at a very high frequency, due to the high-speed rotation of the turbine shaft, and is modified by intermediate electronics 151, which is described in more detail in Fig. 1(b).

Referring to Fig. 1(b), the outputs 122 of the alternator 120 are connected to the inputs 160 (three of them, for a 3-phase alternator) of intermediate electronics, generally designated 151. The first stage of intermediate electronics 151 is an optional transformer stage 162, for boosting the voltage on each of the lines: this ensures, when needed, that there is sufficient dc voltage eventually appearing at the PCU 152 that a complete 240 V sine wave (as per UK mains supply) can be generated at the output of the PCU 152. In certain embodiments, however, the voltage level output by the alternator 120 is high enough such that the transformer stage 162 can be omitted.

Next, the voltages output by the transformer stage 162 at 164 pass to a rectification stage 166, comprising a set of six rectification diodes 168, as is well known in the art. Thus, a rectified, near dc voltage is supplied at outputs 170 of the rectification stage 166, and this, in normal operating conditions appears at the outputs 172 of the intermediate electronics 151.

In the event of a sudden loss of grid connection all alternator load will be lost. This could cause a significant overspeed of the alternator 120, and so as well as a dump valve (not shown) to bypass the turbine, the intermediate electronics 151 includes a safety stage 174 that includes a dump resistor 158 to supply a load to the alternator 120 in the event of loss of grid connection, to prevent overspeed.

A transistor 176 is in series with the dump resistor 158 across the outputs 172, with the base b of the transistor 176 being driven by an overspeed detection unit (not shown). The latter supplies a PWM signal to the transistor 176, the duty cycle of which is proportional to the extent of overspeed, so that the higher the overspeed the greater the load applied by the dump resistor 158.

As can be seen in Fig. 1(b), the power supplied at outputs 172 (referred to herein as dc bus) is a voltage V and current I, and this is supplied to the PCU 152. The PCU 152, which is known in the art, is adapted to convert power from dc to ac at the mains frequency (50 Hz in UK) and voltage (240 V in UK). The PCU in turn is able to vary the dc bus voltage so as to adjust the power output of the system.

Varying the dc bus voltage (V in Fig. 1(b)) in the power conditioning unit 152 controls the speed of the turbine shaft 116. Reducing the bus voltage increases the load on the alternator 120, causing more current to be drawn from the alternator. Conversely, increasing the bus voltage causes the alternator current to drop. By calculating the power (e.g. using P=VI, or using a power measuring device) before and after the bus voltage change, it can be determined whether the power was increased or decreased by the bus voltage change. This allows the point of maximum power output from the alternator 120 to be found and then continually 'tracked' by altering the bus voltage.

In one embodiment, the voltage supplied by the alternator at no load is 290Vac (all voltages are measured line-to-line) on each of the three phases at 45,000rpm, the maximum rated speed of the alternator 120. The lowest speed at which power can be generated is 28,000rpm, at which point the voltage is 180 Vac at no load. Increasing the load will also reduce the alternator voltage: for example at 45,000 rpm the voltage will be 210 Vac at 6.3 kW.

The control of power output by varying the bus voltage may be implemented by suitable analog or digital electronics, microcontroller, or the like. It may also be controlled manually using a personal computer (PC) as the controller 154. Preferably, however, the power output is controlled automatically using a suitably programmed PC or other computing machinery as the controller 154. In either case, the PC communicates with the PCU 152 by means of a RS232 serial communications device, although a RS422 or RS485 adapter could also be used, as is known in the art. The PC may thus, at any time, have a reading of V and I, thereby enabling the instantaneous power to be known.

In the case of automatic PC control, the method of control may be by means of suitable software implementing the following.

```
While system is ON do
Increase bus voltage by one voltage step
Measure new power (=VI)
If new power less than or equal to old power then decrease voltage by one voltage step
        do
        decrease voltage by one voltage step
        measure new power
        while new power more than old power
else    do
        increase voltage by one voltage step
        measure new power
```

while new power more than old power.

It will be appreciated by persons skilled in the art that the size of the voltage step is determined by operating conditions and is a suitably determined small fraction (e.g. 1-2.5%) of the mean bus voltage. In on embodiment, the voltage step change is made about every second.

One other optional feature incorporated in the system is a working fluid purification system, generally designated 170 in Fig. 1. As mentioned hereinabove, if there are non-condensable gases present during the running of the system, overall performance can be substantially reduced, i.e. the pressure ratio of the turbine is lower than it should be. For example, in the turbine mentioned in the examples herein, the input pressure p1 is projected to be 20 bar; and if the output pressure p2 is 2 bar rather than the intended 1 bar, the pressure ratio is 10 rather than 20, giving significantly reduced performance.

A difficulty is that when filling the system initially, the working fluid is a liquid and the rest of the system must be filled with a gas, for example nitrogen. When performing this step the pressure can be reduced to below atmospheric pressure to reduce the mass of nitrogen in the system. However, the pressure cannot be reduced too much, or cavitation will occur in the pump. Therefore, the optimum way to remove the unwanted gas from the system is during the running of the system.

The working fluid purification system 170 includes a conduit 172 connected at one end to a point Q on the second heat exchanger (preheater) 126 and at the other end to control valve 174 which may be at the base entry/exit port 176 of an expansion tank 176, which, in one example, may be the type of expansion tank used in central heating systems. The expansion tank 176 has a flexible membrane or diaphragm 178 so that it may in its lower part contain a variable volume V of gas and/or liquid.

In the example (6kW system) mentioned hereinafter, the measurements are as follows.

System volume	70 litres
Fluid volume	18 litres
Expansion tank volume	50 litres

As can be seen, when the system is initially filled with fluid, there will be 52 litres of nitrogen. Lowering the pressure of this gas with a vacuum pump reduces the amount of gas that has to be held in the expansion tank 176, meaning that it can be made smaller. This pumping also causes the diaphragm 178 expand downwards into the expansion tank, making the whole of the tank, or nearly all of it, available for receiving gases.

As nitrogen gas has a lower density than that of the working fluid vapour, it tends to accumulate at the highest location within the system. At this point (Q in Fig. 1), the fluid can be taken away to the expansion tank 176, the diaphragm 178 allowing expansion to take place, enlarging volume V; i.e., with the control valve 174 open, the gases are allowed to move slowly into the expansion tank 176. As

the nitrogen has a lower density than the working fluid, most of the contents of the expansion tank 176 will be nitrogen, with just a little working fluid.

Once the valve 174 has closed, the expansion tank 176 and its contents cool down naturally, causing the working fluid to condense. The next time the control valve 174 is opened, the (now liquid) working fluid flows back under gravity back into the main circuit of the system (via control valve 174 and conduit 172)), while the non-condensable gases tend to stay in the expansion tank 176 due to their lower density. A cycle of (a) control valve OPEN for a fixed period, followed by (b) control valve CLOSED for a fixed period is used to purify the working fluid, and this cycle may be repeated several times (for example about 3 to 5 times), during the start up of the energy recovery system, to collect as much nitrogen in the expansion tank 176 as possible. In the aforementioned (6kW) system, the control valve 174 is opened for one minute and then closed for ten minutes. The opening and closing of the control valve 174 may be performed manually, or it may be performed automatically by a suitable controller, in this case controller 154.

The system preferably also includes a pressure sensor coupled to the controller 154, the pressure sensor being positioned to sense the pressure at the exit of the expansion device (turbine); and the purification cycle may be repeated if pressure starts to build up during normal running of the system and it is detected at the pressure sensor that the pressure has exceeded a predetermined safe threshold.

Figure 3 illustrates in more detail the coupling of the turbine unit and alternator of Fig. 1(a). Here, the turbine unit is generally designated 114 and the alternator generally designated 120. The turbine shaft rotates about an axis 302 and is integral with a section 304 that forms part of the rotor 306 of the alternator 120. Generally partial cylinder permanents magnets 308 are disposed on the section 304 of the shaft 116. Retaining the magnets 308 in position on the shaft 116 is a retaining cylinder 309: this retaining cylinder (made of a non-magnetic material such as CFRP) ensures that the magnets 308 are not dislodged during high-speed rotation of the shaft 116. The stator 311, incorporating a plurality of windings (not shown) in which current is generated, is mounted around the rotor 306, as is well known in the art, and is enclosed within housing 310. The section 304 of the shaft 116 is supported at one end of the housing 310 by journal bearing 312, and at the other end by the bearing generally designated 314, which is described in more detail hereinafter.

Figure 4 is an enlarged view of the turbine–bearing coupling in Fig. 3. As can be seen, the turbine unit 114 includes a first turbine stage 402 and a second turbine stage 404. High pressure heated working fluid present (at pressure p1) in the space 406 within the turbine unit housing 408 enters via inlet port 410 of the first turbine stage 402 and flows in the direction of arrow F so as to be incident upon a first series of vanes 412 securely mounted on the shaft 116. The fast flowing working fluid thereby imparts rotational energy to the shaft 116. Upon exiting the first turbine stage 402 (at pressure p3), the working fluid flows in the direction of arrows G.

Next, the working fluid at (intermediate) pressure p3 (which is substantially less than p1, but still relatively high) passes, via conduit 413, to the next turbine stage 404. Here, the working fluid enters via inlet port 414 of the second turbine stage 404 and flows in the direction of arrow H so as to be incident upon a second series of vanes 416 securely mounted on the shaft 116. The fast flowing working fluid thereby imparts further rotational energy to the shaft 116. Upon exiting the second turbine stage 404 (at pressure p2), the working fluid flows in the direction of arrow J. Thus, p1 > p3 > p2.

As can be seen, the axial and radial dimensions of the vanes 416 of the second turbine stage 404 are greater than those of the vanes 412 of the first turbine stage 402. In one embodiment, there are two turbine stages of equal diameter, and the axial dimension of the first turbine stage is 3/10 of the diameter, and the axial dimension of the second turbine stage is 4/10 the diameter. In another embodiment, there are three turbine stages. The diameters of the first, second and third turbine stages are in the ratio 4 : 5 : 6. The axial dimension of the first turbine stage is 0.375 x the respective diameter. The axial dimension of the second turbine stage is 0.35 x the respective diameter. The axial dimension of the first turbine stage is 0.33 x the respective diameter.

A washer 418 is provided fixedly attached to a shoulder 420 of the turbine stage 404 and has its other surface abutting a bearing member 422, which is described in more detail hereinafter; and in operation, the working fluid permeates the space between the washer 418 and the bearing member 422, so as to provide lubrication.

The bearing member 422 has a generally T-shaped cross-section. It includes a first bearing surface 424 on a raised portion on the top of the T; and in use, this surface is disposed opposite a second bearing surface 426, of substantially the same annular shape and size, on the shaft 116 near the armature section 304. The bearing member 422 has a central cylindrical channel 428, thereby defining a cylindrical third bearing surface 430 on bearing member 422 that is disposed opposite fourth bearing surface 432 on the outside of shaft 116. A fifth bearing surface 434 is provided on the bearing member 422 on the end thereof opposite the first bearing surface 424, and this is disposed opposite a respective surface of the washer 418. In one embodiment, the working fluid permeates all the spaces defined opposite bearing surfaces 424, 430 and 434 of bearing member 422, thereby providing lubrication of the bearing. In one embodiment, the working fluid is provided as a liquid from the pump 144 (see Fig. 1(a)) via a fluid pipe 160, separate from the main flows, communicating with the outer surface of the bearing member 422.

It will be appreciated that the bearing in this form provides a bi-directional thrust bearing: the bearing member 422 has two bearing surfaces 424 and 434, enabling it to receive thrust in two directions.

Figure 5 shows in more detail the bearing member 422 employed in the bearing in Fig. 4, indicating fluid flows. Figure 5(a) is an end view showing the first bearing surface 424. The flange 502, forming the top of the T, is provided with two screw holes 504 enabling the bearing member 422 to be screwed

or bolted to the housing 310 of the alternator 120. Six equally spaced radially extending first elongate recesses (slots) 506 are disposed in the first bearing surface 424, extending from radial inner extremity of the first bearing surface 424 towards the outer radial extremity of the firs bearing surface 424, enabling the passage of lubricant fluid. As can be seen in Fig. 5(b), each recess 506 does not quite reach the outer extremity 508 of the first bearing surface 424. In the embodiment of Fig. 5(a), each recess 506 is provided with an axially extending second lubrication channels 510, which extend to a circumferential recess (or groove) described hereinafter.

In other embodiments, there may not be a second lubrication channel 510 in each recess 506: for example, Fig. 5(c) illustrates the case where a second lubrication channel 510 is provided in only two of the recesses 506.

Referring to Fig. 5(d), a circumferentially extending recess (groove) 512 is provided in the outer surface 514 of bearing member 422. It can been seen that first lubrication channels 516 (here, four of them, equally circumferentially spaced) extend between the circumferentially extending recess 512 and the interior of the bearing member 422, allowing passage of lubrication fluid. As best seen in Fig. 5(e), the second lubrication channels 510 extend between the first bearing surface 424 and the circumferential recess 512. The ends of the second lubrication channels 510 are also shown in Fig. 5(f). The latter figure also shows a plurality (here six) of second elongate recesses (slots) 516 disposed in the fifth bearing surface 434. Two of the second elongate recesses 516 have second lubrication channels extending therefrom to the aforementioned circumferential recess 512. Figure 5(g) is a partial cross-section showing the recesses and channels in another way.

Returning to Fig. 5(e), the lubrication fluid enters the bearing member 422 in the direction of arrows K. The fluid flows in the direction of arrows L to the first elongate recesses 506 on the first bearing surface 424, in the direction of arrow M to the second elongate recesses 516 on the fifth bearing surface 434, and in the direction of arrow N (into the paper) to the interior of the bearing member and the third bearing surface 430, thereby lubricating the bearing.

Example 1

The specific values for one example (6kW version) of the system are set out below. All pressures are in bar (absolute). All temperatures are in °C. The working fluid is HFE-7100.

t1	t2	t3	t4	t5	t6	t7
180.0	123.4	111.0	165.0	130.0	65.0	55.0

p1	p2	p3
11.5	1.0	3.4

Example 2

The specific values for a second example (120kW version) of the system are set out below. All pressures are in bar (absolute). All temperatures are in °C. The working fluid is hexane.

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Attorney'ref: FP01H04/P-GB

t1	t2	t3	t4	t5	t6	t7
225.0	138.8	123.8	210.0	145.9	74.0	64.0

p1	p2	p3
19.5	1.0	-

The results from the system demonstrate a very useful thermodynamic efficiency (usable electricity out to heat in) for the heat recovery and solar thermal industries — 10% for a source fluid input at 110°C to 22% for a source fluid input at 270°C.

Referring to Figure 6, this illustrates an alternative (magnetic) coupling of the turbine unit and alternator of Fig. 1(a), in another aspect of the invention. The view in Fig. 6(a) is an axial cross-section of the coupling, showing a first rotary member 602 formed of turbine shaft 604 and a first magnetic member 606. In turn, the first magnetic member 606 comprises an armature portion 608, made of steel or iron, integral with the shaft, and a plurality of magnet sections 610, to be described further hereinbelow.

The first rotary member 602 is hermetically sealed inside a housing 612 that contains the turbine (not shown) and working fluid, the housing including a cylindrical portion 614 containing the first magnetic member 606. At least the portion 614 is made of a non-magnetic material, such as stainless steel, nimonic alloy or plastic.

A second rotary member 616 comprises a second shaft 618 and a generally cylindrical second magnetic member 620 integral therewith. The second magnetic member in turn comprises an outer supporting member 622 having a plurality of second magnet sections 624 fixedly attached to the interior thereof.

As best shown in Fig. 6(b), the first rotary member 602 may have a composite containment shell 626 around at least the cylindrical part thereof, so as to maintain the first magnet sections 610 in place during high-speed rotation. The containment shell may be made of a composite such as carbon fibre reinforced plastic (CFRP), kevlar, or glass fibre reinforced plastic (GRP).

Figure 6(c) is a transverse cross-section at A-A in Fig. 6(a). This shows the first magnet sections 610 and second magnet sections 624 in more detail: in this case there are four of each. The magnet sections are elongate, with a cross-section similar to the sector of a disc. The magnet sections are permanent magnets formed of a suitable material, such as ferrite material, samarium cobalt or neodymium iron boron. The direction of the North-South direction for the magnet sections is radial, as schematically illustrated in Fig. 6(d).

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Turning to Fig. 6(e), this shows an alternative embodiment, in which the first magnetic member 606' and the second magnetic member 620' are substantially disc-shaped. The first magnetic member 606' comprises a first mounting section 628 and first magnet sections 610', and the second magnetic member 620' includes a second mounting section 630 and second magnet sections 624'. As before, a non-magnetic portion 614' of the housing (similar to 614 and made of the aforementioned non-magnetic material) separates the faces of the disc-shaped magnetic members 606' and 620', which are in close proximity.

The arrangement of the poles for the magnet sections one or both of the first and second magnetic members 606', 620' is illustrated schematically in Fig. 6(f). As also illustrated in Fig. 6(g), the polarity of the face of the magnet sections 610' (or 624') alternates as you go tangentially from magnet section to magnet section.

These magnet arrangements permit coupling and transfer of rotational energy and torque from the turbine shaft 604 to the shaft 618 of the alternator, and are adapted to do so at relatively high speeds, e.g. 25,000 to 50,000 rpm.

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Claims:

1. A bearing for supporting a shaft rotatable about an axis and at least partially disposed within a housing, comprising:

a bearing member, fixedly attached to the housing and having a first bearing surface, opposite a second bearing surface on the shaft, said first and second bearing surfaces extending generally transverse to the axis, and a cylindrical internal channel defining a third bearing surface extending generally parallel to the axis and disposed opposite a fourth bearing surface on the shaft,

the bearing member including conduits adapted to convey lubricating fluid into at least the space third and fourth bearing surfaces.

2. The bearing of claim 1, wherein the bearing member has a generally T-shaped cross-section.

3. The bearing of claim 1 or 2, wherein the bearing member has, on the end thereof opposite the first bearing surface, a fifth bearing surface extending generally transverse to the axis.

4. The bearing of claim 2, wherein the first surface on the bearing element is defined by a raised annular surface on the top of the 'T' extending partially between the inner radial limit and the outer radial limit of the bearing member.

5. The bearing of claim 3, wherein a plurality of elongate first recesses are provided extending radially in the first surface, thereby facilitating flow of lubricant fluid to the space opposite the first surface.

6. The bearing of claim 5, wherein the first recesses extend partially between the inner radial limit and the outer radial limit of the first surface.

7. The bearing of any of claims 3 to 6, wherein a plurality of elongate second recesses are provided extending radially in the fifth surface, thereby facilitating flow of lubricant fluid to the space opposite the fourth surface.

8. The bearing of claim 7, wherein the second recesses extend partially between the inner radial limit and the outer radial limit of the fifth surface.

9. The bearing of any of claims 2 to 8, wherein at a point between the opposite ends of the elongate part of the 'T'-shaped bearing member, a circumferential recess is defined in the surface at the outer radial limit of the bearing member.

10. The bearing of claim 9, wherein a plurality of first lubrication channels are provided, extending radially between the circumferential recess and the inner radial limit of the bearing member, thereby

permitting flow of lubricant fluid between the exterior of the bearing member and the internal cylindrical channel.

11. The bearing of any of the claims 8 to 10, wherein the bearing member includes a plurality of second lubrication channels, each channel extending axially between a first elongate recess on the first surface and a respective opposite second elongate recess on the fifth surface.

12. The bearing of any of the preceding claims, wherein the number of first and/or second elongate recesses is between 2 and 8, and preferably 6.

13. The bearing of any of the preceding claims, wherein the number of second lubrication channels is between 2 and 8.

14. The bearing of any of the preceding claims, further including a washer, wherein, in use, one surface of the washer abuts the fifth surface of the bearing member and the other surface of the washer is adapted to abut a corresponding surface of a drive element, for example a turbine.

15. The bearing substantially as hereinbefore described with reference to the accompanying drawings.

16. A energy recovery system, for extracting energy from a source of waste heat, the system being a closed system with a circulating working fluid, comprising a heat exchanger, an electromechanical conversion unit, a cooling system and a turbine unit, the heat exchanger supplying, in use, the working fluid to said turbine unit as a gas, wherein the turbine unit is mechanically coupled to the electromechanical conversion unit via a shaft, the shaft being supported by a bearing according to any of the preceding claims.

17. The system of claim 16, further including a secondary working fluid supply line from the cooling system to the bearing whereby working fluid is supplied to the exterior of the bearing member, thereby providing the lubricant fluid for said bearing.

18. The system of claim 17, wherein the working fluid is supplied to the bearing as a liquid.

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Attorney'ref: FP01H04/P-GB

Abstract

A sliding **Bearing** for supporting a shaft

A bearing for supporting a shaft rotatable about an axis and at least partially disposed within a housing, comprises ~~comprising;~~ a bearing member 422, fixedly attached to the housing and having a first bearing surface 424, opposite a second bearing surface 426 on the shaft 116, said first and second bearing surfaces 424, 4 extending generally transverse to the axis, and a cylindrical internal channel 428 defining a third bearing surface 430 extending generally parallel to the axis and disposed opposite a fourth bearing surface 432 on the shaft 116 the bearing member including conduits adapted to convey lubricating fluid into at least the space between the third and fourth bearing surfaces. In one embodiment, the bearing member 422 has, on the end thereof opposite the first bearing surface, a fifth bearing surface 434 extending generally transverse to the axis: this advantageously provides a bi-directional thrust bearing, as the bearing can take thrust in two directions. Several channels and slots enable lubrication fluid to pass from the exterior of the bearing member to all of the bearing surfaces. The bearing is particularly suited to compact high speed applications, e.g. in coupling an alternator to a high speed turbine.

(Fig. 4)

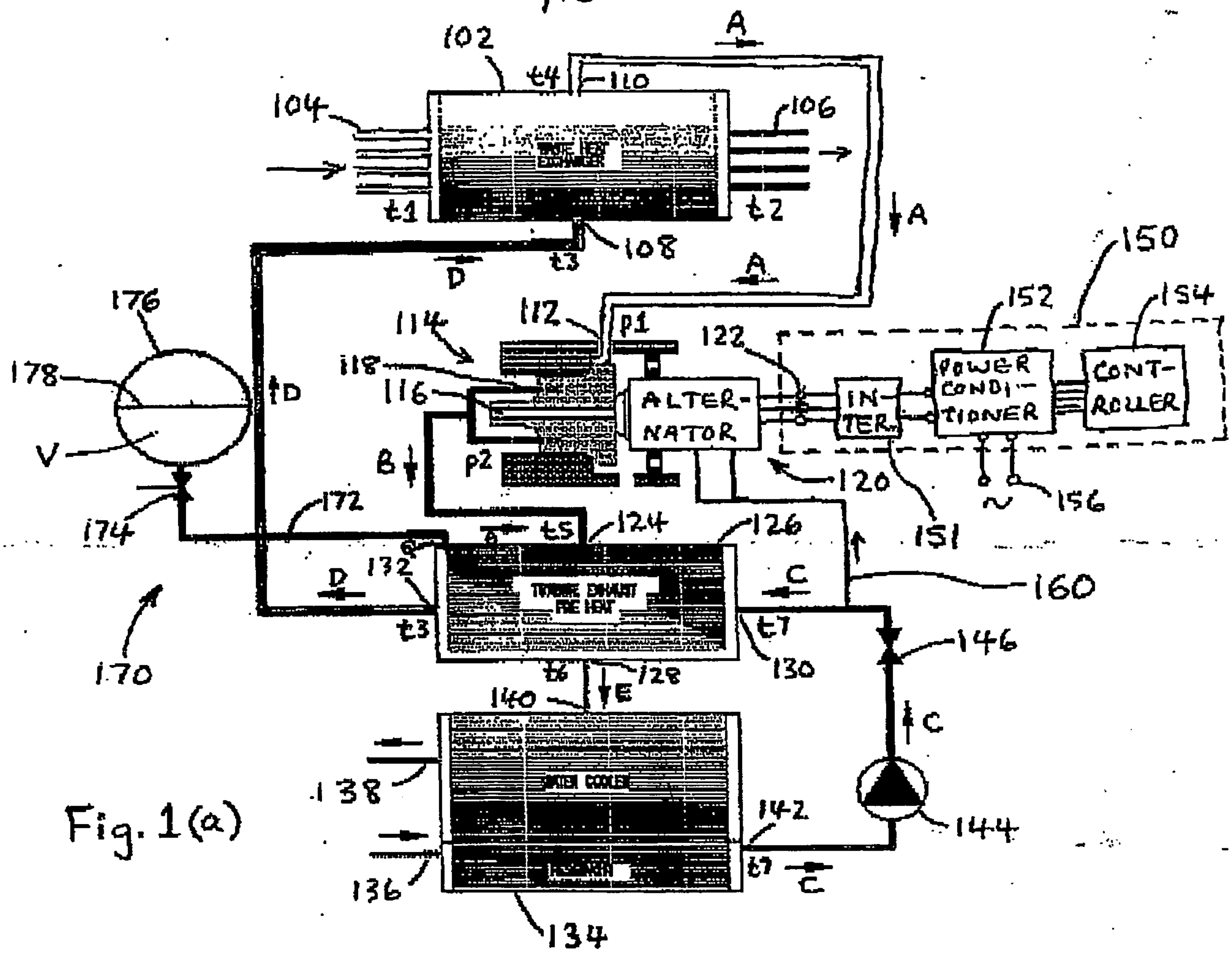
102
t4
110
A
104
106
t1
t2
A
108
t3
D
A
150
176
114
112
p1
122
152
154
178
118
116
D
ALTER-
NATOR
IN-
TER.
POWER
CONDI-
TIONER
CONT-
ROLLER
V
B
p2
120
156
151
174
172
124
126
t5
TURBINE EXHAUST
PRE HEAT
D
132
C
160
t3
t7
146
170
t6
128
130
140
E
C
138
WATER COOLER
142
144
136
t7
C
134

Fig. 1(a)

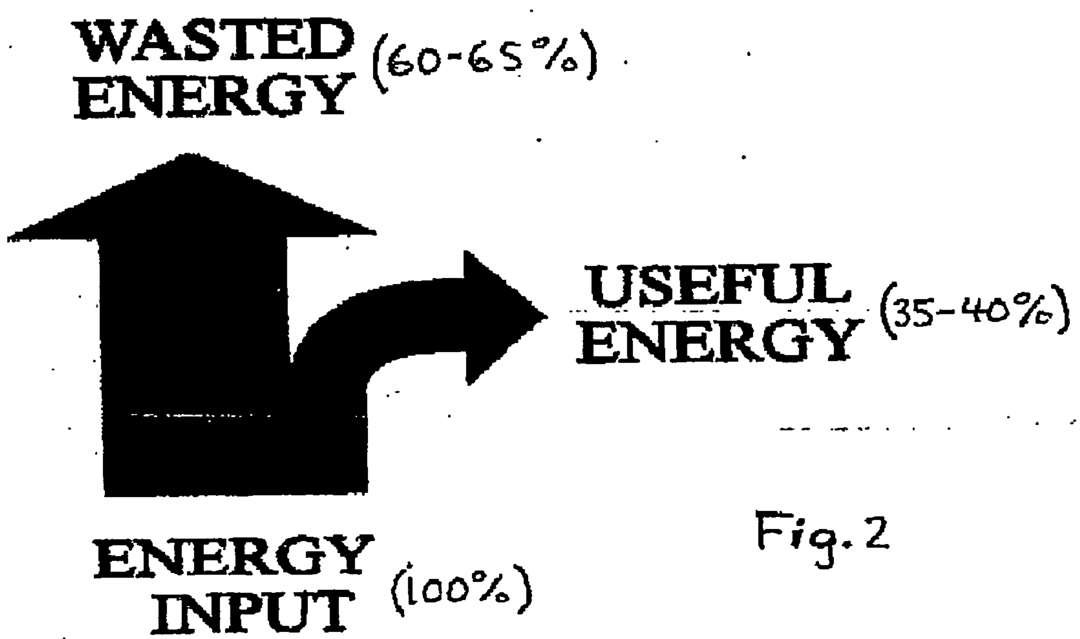
WASTED
ENERGY
(60-65%)
USEFUL
ENERGY
(35-40%)
ENERGY
INPUT
(100%)

Fig. 2

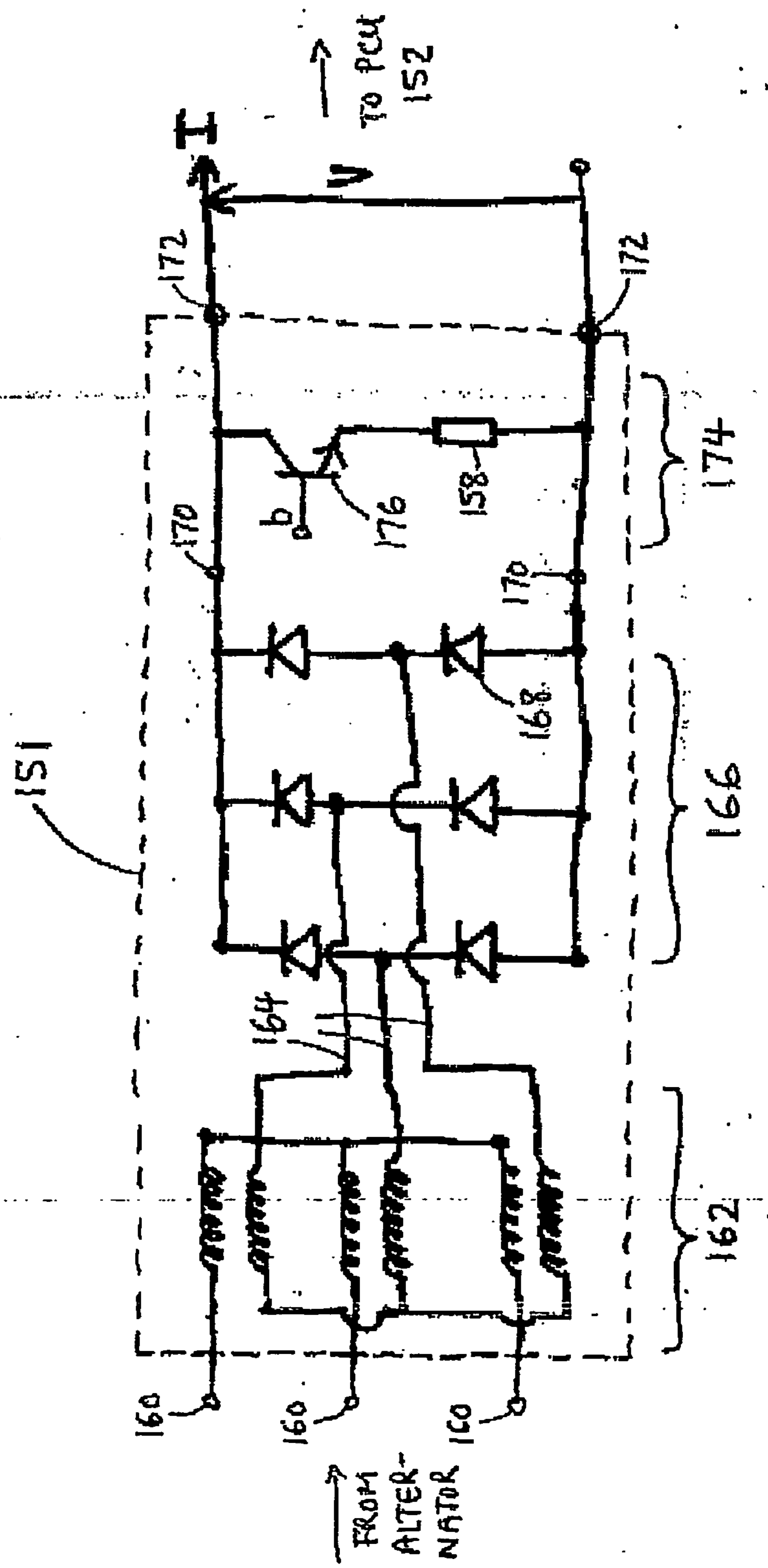
TO PCU 152
I
V
172
172
170
170
b
176
158
174
168
166
151
164
162
160
160
160
FROM ALTER-NATOR

Fig.1(b)

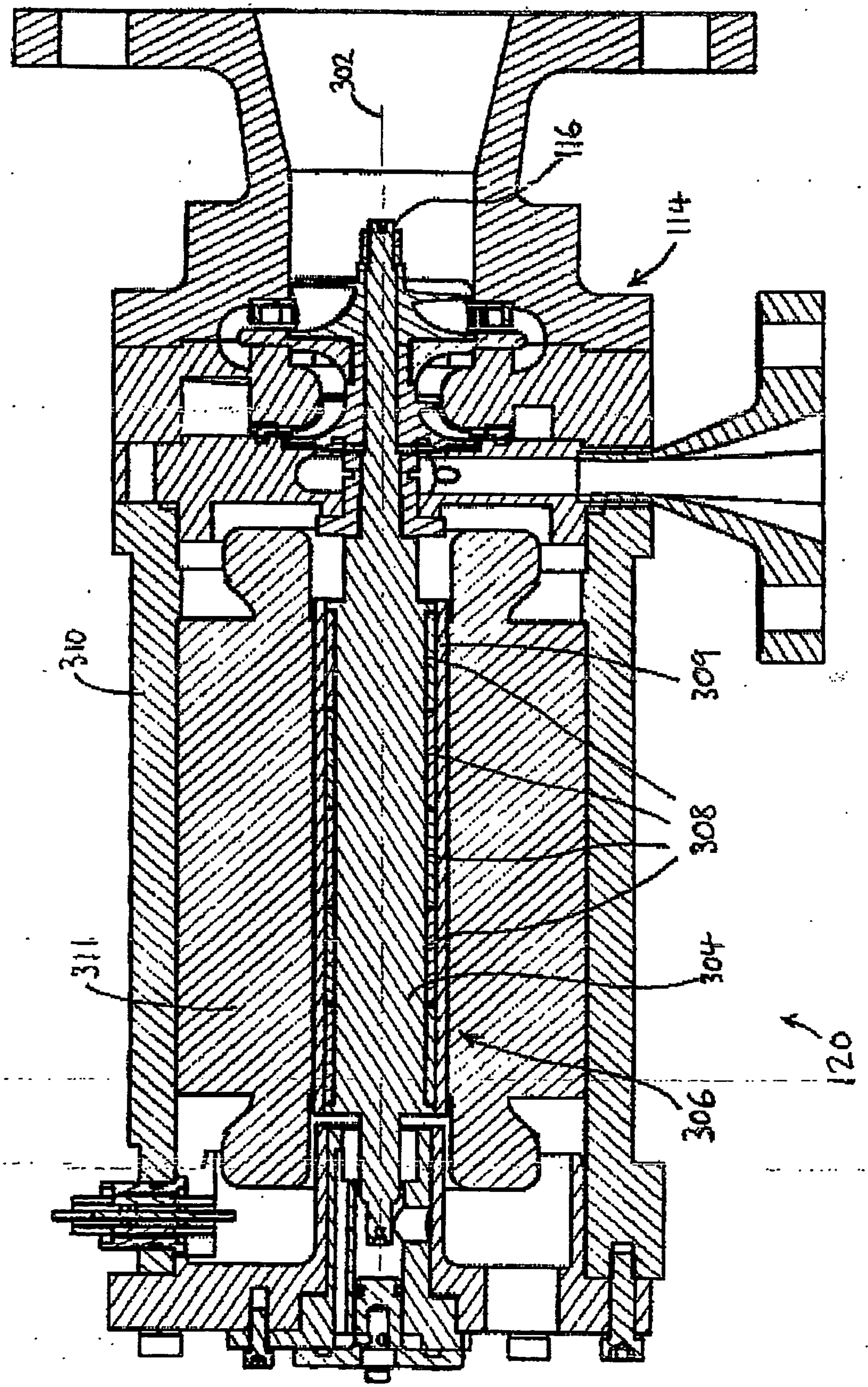

Fig. 3

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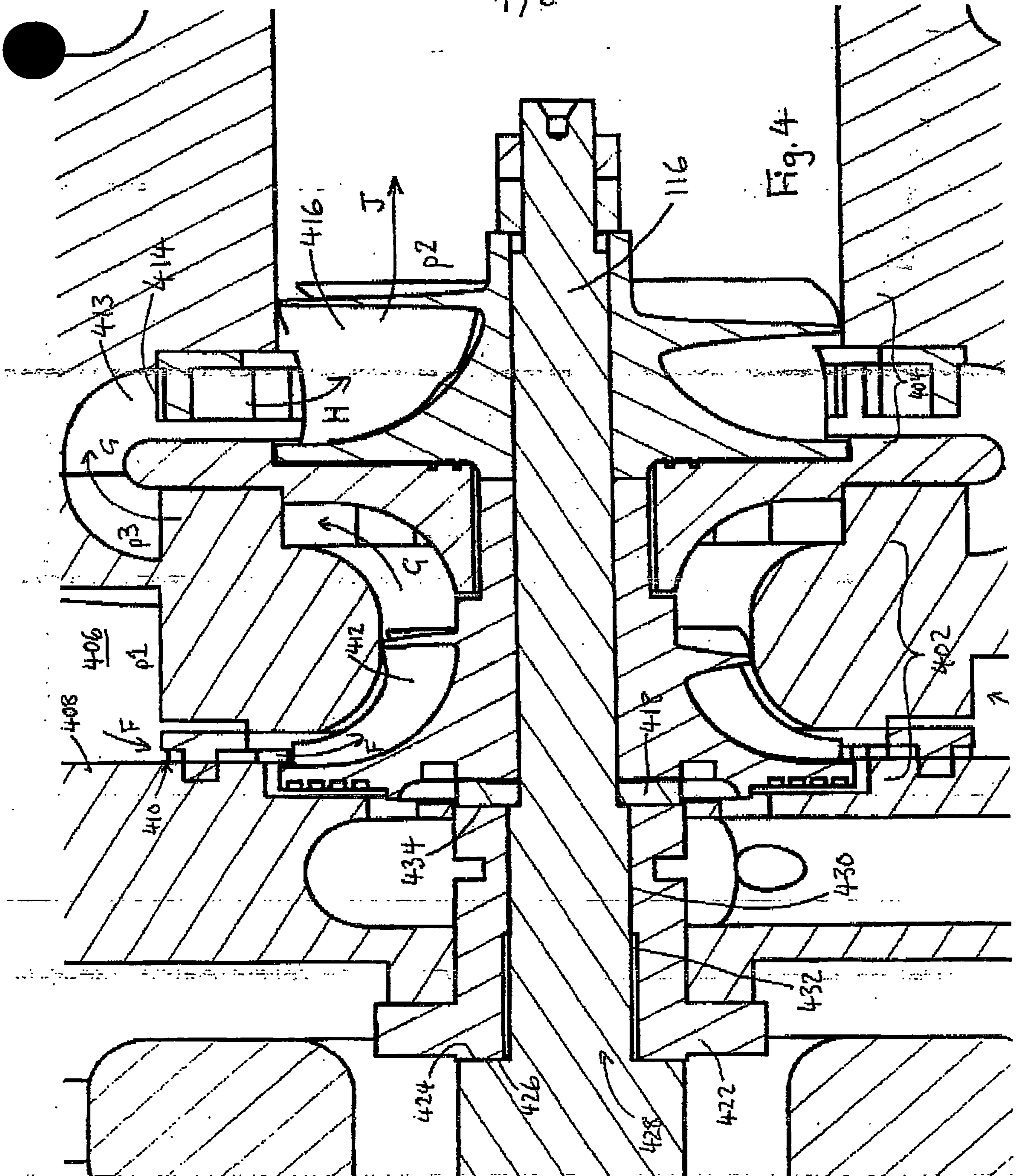
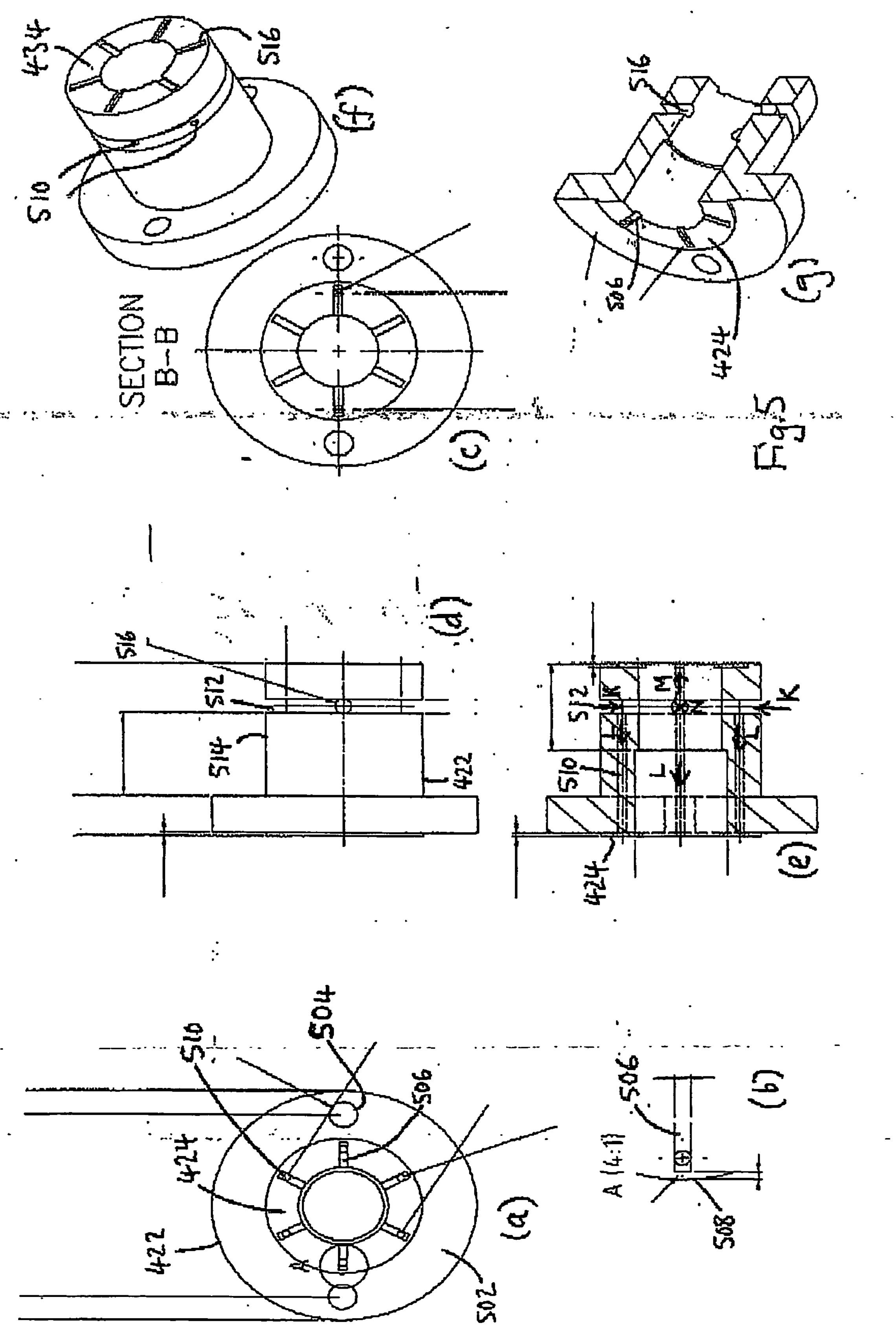
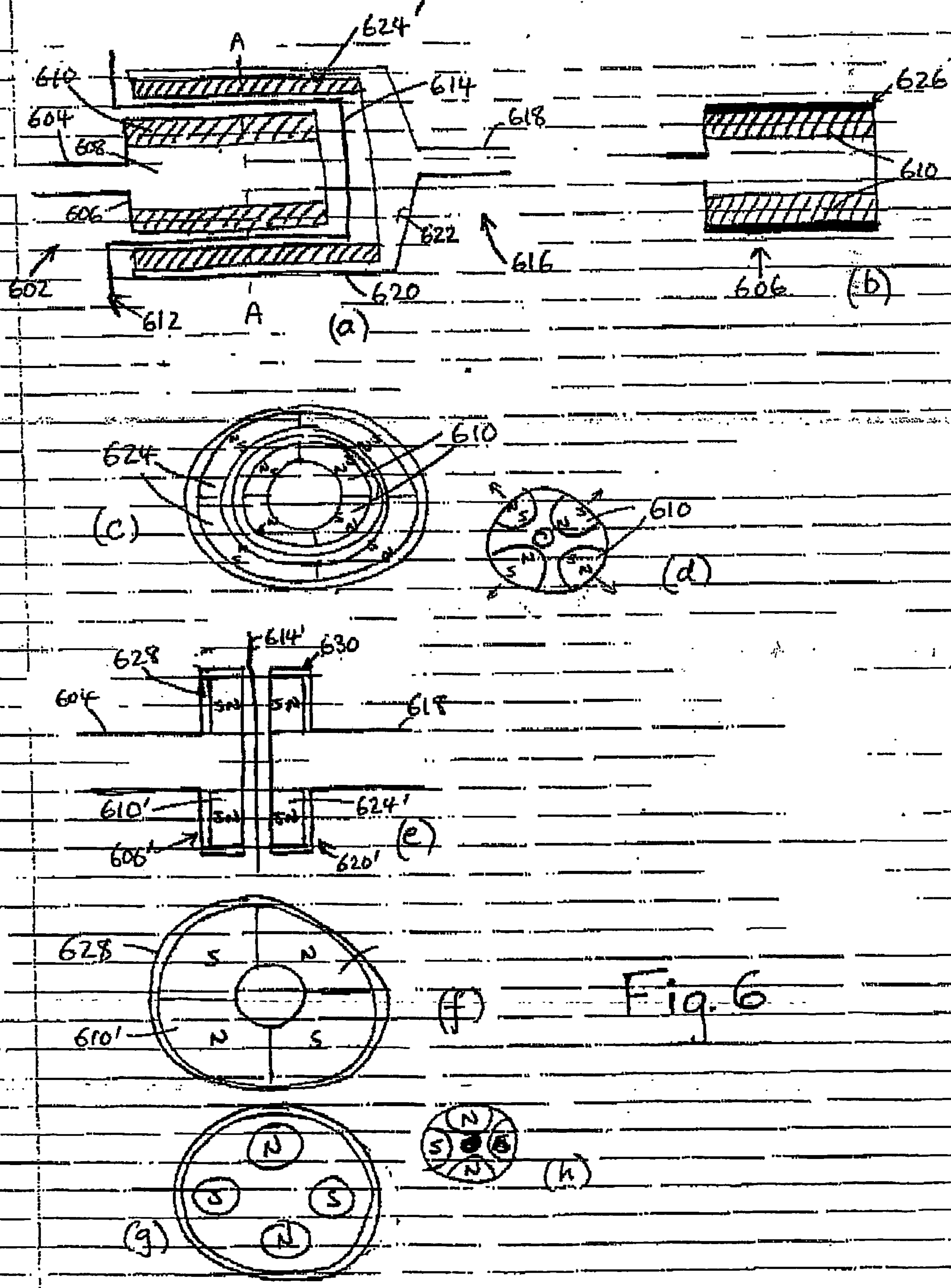


Fig. 4
116
416
J
p2
413
414
H
G
p3
406
p1
408
F
410
412
418
404
402
434
430
432
424
426
428
422


434
516
(f)
510
SECTION B-B
(c)
516
506
(g)
424
Fig 5
(d)
516
512
514
422
510
512
K
M
N
L
(e)
424
504
510
506
424
422
502
(a)
A (4:1)
506
508
(b)

6/6



A
624
610
614
604
618
608
606
622
616
602
620
612
A
(a)
626
610
606
(b)
610
624
(c)
610
(d)
614'
630
628
604
618
610'
624'
606'
620'
(e)
628
S
N
610'
N
S
(f)
N
S
S
N
(g)
N
S
S
N
(h)

Fig. 6

PCT/EP2004/009580